DESIGN CONSIDERATIONS FOR BEARINGLESS ROTOR HUBS

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Abstract

Five studies were performed under contract for concept definition of the Integrated Technology Rotor/Flight Research Rotor (ITR/FRR) Project. The purpose of these studies was to examine numerous hub design concepts with the potential of meeting the technical goals and specifications set for the ITR/FRR. Thirty-three concepts were proposed, including 21 bearingless designs. Although the level of effort in these studies was not equivalent to a preliminary design, many important design issues for advanced rotor hubs were examined, particularly with respect to bearingless-rotor designs. The critical aspects of bearingless-hub design are summarized here; they include the design of the flexbeam, the design of a torque structure for pitch control, and the means of augmenting rotor lead-lag damping. Representative designs from the concept definition studies are used to illuminate these design aspects. The capability of the designs to meet the ITR/FRR technical goals is also discussed.

Introduction

The structural simplicity of a helicopter rotor that allows the centrifugal loads, as well as all flapping, lead-lag, and torsional motions, to be carried by flexible structural elements rather than by rolling-element or elastomeric bearings is attractive to the rotor designer. The development of composite materials in recent years has provided the necessary conditions for the design of rotors without bearings, but the apparent structural simplicity of a bearingless rotor conceals a number of difficult design problems, including the strain on a flexbeam element caused by combined deformations, aeromechanical stability, blade-pitch control, and redundant structural load paths.

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The first major effort to develop a bearingless rotor was a matched-stiffness rotor installed on the XH-51A by Lockheed-California for the U.S. Army Applied Technology Laboratory at Fort Eustis, Va., in 1966.[1] This development was only partially successful and, in a sense, was premature because of our limited knowledge of aeromechanical stability and of the use of conventional materials. The first successful efforts to apply bearingless rotor technology were made in the design competition for the Utility Tactical Transport Aerial System (UTTAS) in the early 1970s, in which both the Sikorsky YUH-60A (Ref. 2) and the Boeing Vertol YUH-61A (Ref. 3) used stiff inplane bearingless designs for their tail rotors. The Sikorsky design was carried on to the production UH-60A aircraft and has also been used successfully for the S-76 helicopter. In many ways the application of this technology to tail rotor design has been a simpler task, and these efforts have continued with prototype efforts at Messerschmitt-Bölkow-Blohm (MBB)[4] for a replacement tail rotor for the BO 105 and BK 117 aircraft, and at Hughes Helicopters[5] with the Composite Flexbeam Tail Rotor (CFTR) for the AH-64A. Although not strictly a bearingless design, prototype teetering rotors using flexbeams for chordwise and torsional motion have been built and tested at Kaman[6] and Bell Helicopter Textron.[7]

The design of a bearingless main rotor, however, remained a difficult problem. By the mid-1970s sufficient work had been done so that the U.S. Army decided again to investigate the feasibility of a bearingless design, particularly with respect to aeromechanical stability. The Applied Technology Laboratory selected Boeing Vertol to build the Bearingless Main Rotor (BMR) for the BO 105 in 1976. The BO 105/BMR first flew in 1978 with a flexbeam made of back-to-back C-beams and with blade pitch that was controlled with a torque tube.[8,9] Other manufacturers such as Aerospatiale, with the Triflex design,[10] and Bell Helicopter Textron, with the Model 680 rotor[11] have pursued bearingless designs.

The bearingless-rotor development efforts to date have been successful in demonstrating that the critical design problems with bearingless rotors can be overcome. However, with each of these development efforts, success in solving a particular design problem has usually been achieved at the expense of another design goal. For example, long fatigue life may be achieved but only by compromising the desired hub-moment stiffness goals. Before bearingless-hub technology can be considered to be sufficiently developed for application to new systems, it must satisfy a number of important design attributes simultaneously. These attributes include specified weight, drag, fatigue life, hub-moment stiffness, or equivalent flap hinge offset, stability, maintainability, ballistic vulnerability, and cost.

In the context of the entire rotor system it is desirable that the advances resulting from a bearingless-hub design be integrated with such technical advances as those in aerodynamics, structures, acoustics, and vibration, for the entire rotor system. The next generation of rotorcraft requires a rotor system that integrates the advances of the last decade in the areas of hub design, materials, aerodynamic performance, reliability and maintainability, and acoustics and demonstrates that these advances can be achieved at low cost, low weight, and low technical risk. In view of this need, the U.S. Army Research and Technology Laboratories (AVRADCOM) have joined with NASA Ames Research Center in a joint program to develop an Integrated Technology Rotor/Flight Research Rotor (ITR/FRR). The objective of the ITR program is to make significant advances over a broad spectrum of technologies; the FRR will be a derivative of the ITR and will have substantial research variability that will be tested on the NASA/Army Rotor Systems Research Aircraft (RSRA). To meet the ITR objectives, a set of technical goals has been established that range from rotor blade performance attributes, such as figure of merit and rotor equivalent L/D, to such hub characteristics as hub-moment stiffness and hub drag. In each case, the technical goals act as significant technology drivers, and collectively they define a rotor that is a substantial advancement of the state of the art.

The ITR/FRR Project is structured in four phases: predesign studies, preliminary design, detail-design/fabrication, and test. The predesign-studies phase is composed of three efforts and is nearly complete. The first effort is a comparison of the industry mathematical models for stability prediction with a variety of hingeless and bearingless rotor experimental data sets. These studies provide an assessment of existing methods in this critical area. The second effort is the concept-definition studies[11-16] which have examined various hub concepts that can meet the ITR/FRR technical goals; it is the subject of the present paper. The final effort is the FRR predesign studies, of which two are complete;[17,18] the remaining two will be completed in the near future. The second phase of the project, the preliminary design of the ITR/FRR is currently under way.

This paper will discuss the results of the five concept-definition studies, with particular emphasis on hub design features that are believed to be critical to the success of the ITR/FRR. The objectives of the concept-definition studies will be discussed first. The hub concepts proposed in these studies will be categorized by the design approaches used to provide the necessary flapping, lead-lag, and torsional flexibility. Design considerations that are particularly critical for bearingless designs, including the flexbeam, the pitch-control structure, and the means of achieving an aeromechanically stable configuration, will be addressed. The paper concludes with a comparison of selected concepts with the ITR/FRR technical goals.

ITR/FRR Concept-Definition Studies

Central to the technologies on which the ITR/FRR is based are those related to the design of the hub. For this reason, the concept-definition studies were undertaken to examine various hub concepts that have the potential of meeting the ITR/FRR technical goals. The level of effort involved was less than would be expected in a preliminary design. In part, the objective was to examine as many candidate hub concepts as possible and in this way better understand the design problems that will be encountered in subsequent phases of the project.

A set of technical goals and specifications for the hub was established to guide the concept-definition studies. The technical goals of the ITR/FRR Project were used as a base, and those goals that applied to the hub design were used as specific objectives for these studies. The technical goals for the hub design are given in Table 1. Basically, these goals represent the technology drivers for the hub design. The flat-plate drag area is a significant reduction from current technology, but may be possible with a small frontal area and hub fairings. The hub weight goal is also difficult to attain.

An important objective in reducing cost and improving maintainability is to reduce hub parts count. Some current production designs have more than double the design goals of 50 parts. The hub-moment stiffness goal of 2,600 ft·lb/deg is quite low and may significantly affect potential hub concepts. Assuming no stiffness in the flap bearing or flexbeam, this is roughly equivalent to a 4.2% hinge offset and is lower than the 4.7% offset of the articulated UH-60A. A low hub-moment stiffness is desired to improve vibratory characteristics, gust response, and handling qualities. Although a higher stiffness may be adequate for some applications, this can be easily obtained once the technology for low stiffness is demonstrated. However, the converse is not true. The allowable tilt angle, fatigue life, and mean time between removal (MTBR) have all been set high as a means of pushing the technology to improve the reliability and maintainability of the design.

Some of the goals are complementary. Clearly, a design that minimizes size and weight will also reduce hub drag. Simplicity in design will reduce the number of parts and probably the MTBR as well. However, the goals for hub-moment stiffness, tilt or flapping angle without fatigue damage, and fatigue life show a different relationship. For a very stiff hub, it is expected that the allowable flap angle before fatigue damage occurs will be relatively small. However, a stiffer hub will not require much blade flapping so the fatigue life may or may not decrease. A very soft hub will have a large allowable tilt angle, but its required flapping will be significantly increased and the effect on fatigue life is unclear. Other goals, although not quantified, were established to 1) incorporate provisions for auxiliary dampers, 2) maintain swashplate actuator loads at current levels, and 3) design for low production cost. Specifications for the hub design were made as well, and these act as design constraints as opposed to technology drivers. For example, specifications requiring a four-bladed design, manual blade-folding, and low vulnerability do not push technology but significantly affect what the final design looks like.

The studies required each company to examine at least five hub concepts and, based on their examination, to select at least two designs that would best meet the project technical goals. The selected concepts were then developed further and compared with each other on the basis of a government-supplied merit function. Additional tasks within the studies addressed potential FRR concepts and modifications required for the RSRA. The study contracts were with Bell Helicopter Textron, Boeing Vertol, Hughes Helicopters, Kaman Aerospace, and Sikorsky Aircraft; they are reported in Refs. 12-16.

Hub Concepts Classified by Flexibility

Under normal operating conditions, a rotor blade will undergo flapping, lead-lag, and torsional motions. The means by which these motions are accommodated, whether by rolling-element or elastomeric bearings, or by flexible structural elements can be used as a system of classification, that is, whether the rotor is an articulated, hingeless, or bearingless design or some combination of these. As with any system of classification there may be concepts that do not easily fit in one category or another. The intent of the classification systems used in this paper is to afford a better understanding of the design problems involved in developing a hub for the ITR/FRR, not to develop a rotor-hub taxonomy.

In their concept-definition studies, the five companies were asked to propose at least five hub concepts that would have the potential of meeting the ITR/FRR technical goals. Thirty-three concepts were proposed.[12-16] In Fig. 1, these concepts are classified accordingly as the primary flap, chord, and torsional motions are accommodated by articulation, such as elastomeric bearings, or by structural elements. In the case of flapping, a further distinction is made on the basis of whether flapping is provided for with a conventional, articulated hinge or with a gimbal design. Some current production and prototype rotors are also classified in Fig. 1 for illustrative purposes.

In Table 2, designation codes are assigned to the 33 concepts; these codes are used throughout the succeeding discussions. Source information, nomenclature, and corresponding figure numbers (in the cited references) are also provided in Table 2.

Baseline Concepts

In examining a variety of hub concepts, some of the manufacturers included an advanced technology derivative of a current production design. In this way, they were able to examine how well updated concepts could meet the ITR/FRR goals without going to the increased risk inherent with bearingless or other more novel designs. Examples are concepts SA1, which is an improved technology UH-60A composite hub; BH7, which is a composite hub derivative of the Model 412; BV5, which is a Boeing Vertol variant of the Aerospatiale

Starflex hub; and HH8, an advanced technology AH-64A hub. As an example, Fig. 2 is a sketch of the SA1 hub without its single-element elastomeric bearings. Its use of composite materials would increase reliability and decrease weight, but it would still require lag dampers and their attendant complexity.

Articulated Flap Hinge Designs

Two designs were proposed in which a conventional articulated flapping hinge was used in conjunction with a lag-torsion flexure (BH6 and BV6). An example (Fig. 3) is the BH6 concept which places an elastomeric flap hinge as close to the hub centerline as possible, with a flexbeam element outboard that allows the lead-lag and torsion motions. The pitch cuff has a root shear-restraint that incorporates an elastomeric damper. Both of the combination articulated-flexbeam designs were attempts to lower the hub-moment stiffness to meet the goal of Table 1 (2,600 ft·lb/deg).

Gimballed Designs

Four gimballed design concepts were identified by three of the companies. The KA4 concept was novel in that it used a composite flat plate or shell to provide the gimbal freedom; in the other concepts (BH5, SA3, and SA6) elastomeric bearings were used. In concept SA6, which has been previously studied,[13] hub-moment stiffness is controlled with a separate gimbal spring (Fig. 4). As with the articulated flap hinge, the primary reason for going to the gimbal design is to achieve very low hub-moment stiffness.

Bearingless Designs

Twenty-one of the 33 hub concepts from the concept-definition studies were bearingless designs. Many different design approaches were tried. Flexbeam designs ranged from single-element flexbeams with relatively little radial variation in the cross section to designs with the cross sections highly tailored along the length of the flexbeam. Multiple-element beams (two, four, and six elements) were proposed, as were laminated beams and designs that used shoes to constrain flap bending. The torque structures used to control blade pitch showed less variety; for the most part they either enclose the flexbeam (pitch cuffs) or are separate from the flexbeam (torque tubes). The flexbeam-to-hub attachment is also a difficult design area and one that received considerable attention.

Flexbeam

Design Considerations

The hub weight and drag technical goals will require that the design be as light and compact as possible. In turn, the limits in reducing hub weight and size will be defined by stiffness requirements and material strengths. A certain minimum cross section is required for the flexbeam to support the blade centrifugal loads. Under normal operating conditions, blade deformations will require additional material in the flexbeam to accommodate the bending stresses, and this increases stiffness and causes the equivalent flap hinge to move outboard. The lead-lag flexbeam stiffness will be governed by the need to place the rotor lead-lag frequency at about 0.7/rev. If the flexbeam is too soft in chordwise flexibility, the rotor will be more susceptible to aeromechanical instability; if it is too stiff, the chordwise loads will increase because of the 1/rev resonance. For torsional motions, it is necessary to minimize the torsional stiffness so as not to increase actuator loads over current designs.

Depending on the details of the flexbeam design, critical loading conditions may be due to low-cycle/high-strain loading, or high-cycle/low-strain loading. In the former case, the flapping and lead-lag loads that occur during startup and shutdown conditions may size the flexbeam. As hub-moment stiffness is reduced to meet the technical goal, the ability of the flexbeam to carry the static droop loads is decreased until the droop-load condition becomes critical in sizing the flexbeam. If droop stops are incorporated in the design, then they can be used to take the static droop loads, but it is not clear that this can be done without penalties in parts count, hub weight, or hub drag. Rotor startup or stop loads may also be critical for sizing the flexbeam in the chord direction.

High-cycle/low-strain loading will occur because of 1/rev oscillatory flap and torsional loads in forward flight. A minimum flapping angle of ±5° without fatigue damage has been specified as a goal and this, in conjunction with the fatigue life goal, may have a critical effect in determining the flexbeam flapping capability. Similarly, the equivalent endurance chord loads may also be critical in sizing the flexbeam in the chordwise direction. The cyclic torsional moments required to control the rotor in forward flight require twisting of the flexbeam of the order of ±15°, and this is also a critical high-cycle/low-strain loading.

The effects of combined loadings must also be considered, although, in general, the maximum flap loads will be 90° out of phase with 1/rev chord and torsional loads so that the combined condition is not especially severe. This is not the case, however, for Coriolis-induced 2/rev chord loads, and the combined loading condition may indeed be critical.

The critical flap and chordwise loading conditions for the hub design can be determined largely from experience with past designs. With the use of composite materials, a great many different ways of accommodating these loads are open to the designer. Considerable work still needs to be done with composites to develop their utility, including an increased allowables data base and the use of hybrid or laminated composites, but by and large this effort is tractable. This is not so for vulnerability considerations, where it is significantly more difficult to design for a specified threat with assurance that the design when tested will have the required vulnerable area. The uncertainty in this aspect of the design process leads to conservative design approaches, and in this sense vulnerability considerations are the indeterminants in the flexbeam design process – all of the design considerations that have been discussed for the sizing of the flexbeam may be obviated by vulnerability requirements.

<u>Design Approaches</u>

The 21 bearingless-hub concepts proposed in the concept-definition studies are classified on the basis of the flexbeam design characteristics shown in Fig. 5. There is a degree of arbitrariness in these classifications, but, as before, the purpose of the system of classification is to provide a convenient framework within which the design approaches that can achieve the ITR/FRR goals can be understood. The approaches are classified according to whether: 1) they use a single-element flexbeam (as for the UH-60A tail rotor), a multielement beam (as on the BO 105/BMR), or a laminated beam; 2) the flexbeam cross section is tailored along its length to separate flap flexibility from lag and torsion (as on the Bell Helicopter Textron 680 rotor); and 3) a shoe is used to control flap-bending curvature.

<u>Cross-Sectional Tailoring of Flexbeam</u>. For a flexbeam of uniform cross section, the peak flapping-strain will be at the root end of the flexbeam, and the chord and torsion strains will be relatively constant along the length of the flexbeam. By tailoring the cross section of the flexbeam along its length, it is possible to separate the maximum strains into a flapping flexure inboard and a lag-torsion flexure outboard. This has the advantage of simplifying the flexbeam design in that the flap and lag-torsion flexures may be individually sized for separate load criteria and for reducing the hub-moment stiffness. The disadvantages of tailored cross-section flexbeams are that they may be difficult to manufacture and there may be structural difficulties where one cross section transitions to another, or changes too rapidly.

An example of a uniform-cross-section approach is concept SA2 (Fig. 6). For this design, an optimization program was used[16] that varied the cross-section flange and web thicknesses to establish the correct lead-lag frequency at minimum weight. Loading conditions included oscillatory flap and chord bending moments but no torsion. The cross section was not changed over the span, although the web and flange thickness tapered outboard. The geometry is very simple, and, although the hub-moment stiffness was not minimized as part of the optimization process, the calculated value with a graphite flexbeam of 3,050 ft·lb/deg is not unreasonably high.

An approach that tailors the cross section in a manner similar to that of the Bell Helicopter Textron Model 680 is concept BH2 (Fig. 7). The inboard cross section is rectangular and is relatively stiff chordwise and in torsion, but soft in flapping. The outer section transitions to a triple-H cross section which is stiff flapwise, but relatively soft in lead-lag and torsion. The thickness distribution of the flap flexure segment has been determined using a formal optimization procedure[illegible] in which only flap degrees of freedom are considered. Based on flapping of ±5°, a thickness distribution was calculated that would minimize flexure thickness and, hence, hub-moment stiffness. For this concept, in which fiberglass is used, the resultant hub-moment stiffness was 3,170 ft·lb/deg. It is particularly interesting that in comparing concepts SA2 and BH2, different design approaches, loading conditions, optimization functions, and constraints were used, but the resulting hub-moment stiffnesses are similar. Although detailed calculations might show more differences than are indicated here, this suggests that a minimum value of hub-moment stiffness for single-element flexbeams may be achieved using a number of alternative design approaches.

<u>Multiple-Element Beams</u>. Flexbeam designs may have two, four, or more elements, and in some cases this may be advantageous. The BMR tested on the BO 105 aircraft had a dual-element flexbeam, which

was useful both in reducing the torsional rigidity and in accommodating the torque tube used for pitch control. An example of a design with four flexbeam elements is concept BV1 (Fig. 8). The four elements are interleaved through the hub; that is, one blade pair passes over the other blade pair and they extend out to and pass around the single blade-attachment bolt. In this case, the rationale for a four-element beam is to reduce hub-clamp forces and perhaps reduce vulnerability. In a sense this concept is like the original BO 105/BMR, with the webs of the C-beams removed to make the flexbeam/blade and flexbeam/hub attachment more efficient. Unlike the BMR, however, the thickness, width, and spacing of the individual beams are varied to improve the flexbeam characteristics. The hub-moment stiffness of 4,200 ft·lb/deg is considerably higher than the goal, as is to be expected with a multiple-element beam of this configuration. Other concepts proposed show considerable variety in the number of elements selected, and in some cases the flapping and lag-torsion flexures have different numbers of elements.

<u>Laminated-Element Flexbeams</u>. The laminated-element flexbeam is attractive as a means of relieving flexbeam stresses caused by blade flapping. This approach has been successfully used for rotor designs, most notably in the metal strap-pack on the OH-6A and AH-64A aircraft. In applications proposed in the concept-definition studies, the laminates would be composites and the shims between the laminates would be a Teflon-like material or elastomer that does not transmit a significant shear. Laminated elements have the beneficial effect of reducing flapping stresses, but they also reduce the flapping stiffness so that, in general, they must be used with a shoe to control the bending curvature at the root of the laminated flexbeam. (It is noted, however, that concept BH4 is sufficiently stiff that a shoe is not required; on the other hand, concepts BV4, KA1, KA3, and SA5 require a shoe to control the root-bending curvature even though they are of solid construction.)

An example of a laminated-element flexbeam is concept HH1 (Fig. 9). This approach is tailored, and only the inner end of the flexbeam is laminated. Outboard of this section, the flexbeam transitions to a flattened X cross section which controls the chord and torsion behavior of the flexbeam. By going to the laminated design, the hub-moment stiffness is reduced significantly below that achievable with a solid flexbeam. For this concept, the calculated hub-moment stiffness is 1,830 ft·lb/deg, well below the goal.

In concept KA2 (Fig. 10) dual-element, laminated flexbeams are used. This approach is untailored in the sense that all the flapping, lead-lag, and torsional motions are taken up in the dual-element flexbeam. For the design shown here, in which the pairs of laminated flexbeams carry through the hub center, an additional useful property of the lamination occurs. Because the laminates are essentially clamped at the blade attachment and free at the hub center, the reduction in hub moment occurs for cyclic flapping, which is desired, but not for collective flapping (coning). Thus, the design has significantly less static droop than a single-element flexbeam of comparable hub-moment stiffness. Calculations for two-ply and five-ply laminated-element beams show hub-moment stiffness ranging from 690 to 1,440 ft·lb/deg.

<u>Lag-Torsion Deformation Requirements</u>. The lag-torsion flexure is designed both for lead-lag frequency placement and to minimize the torsion moment caused by blade twist and, hence, control loads. Within the flexbeam, the moment caused by twist depends on the shear rigidity, GK, where G is the material shear modulus and K is the cross-sectional inertia term; on cross-section warping and differential bending effects; and on centrifugal stiffening effects that depend on the cross-section radius of gyration.[illegible] The ratio of the shear modulus to the bending modulus for unidirectional composites is significantly lower than for isotropic materials, and this provides the designer with more flexibility. But despite these materials advantages, the cross section must be carefully selected to reduce the twisting moment. Figure 11 shows the cross sections of the lag-torsion flexures of some of the concepts that have been described. The variety of design approaches suggests that there is no true optimum cross section. It is not enough to minimize the torsion moment under the lead-lag frequency constraint; in addition, the resulting shear stresses must be within allowables and the transitions from the lag-torsion flexure cross section to the flap flexure, blade, or hub attachment cross sections must be feasible from a manufacturing viewpoint.

The cross section for concept KA2 has a large radius of gyration, which is inherent in the V-frame type of layout, as shown in Fig. 10. But because the two flexbeam elements meet in an apex at the blade attachment, the torsional moment owing to centrifugal force, the "trapeze" effect, is significantly reduced. In addition, the use of laminated sections acts to reduce the shear rigidity of the separate elements.

Pitch-Control Concepts

Design Considerations

A pitch-control or torque structure of some kind is required to twist the blade outboard of the flexbeam to establish the blade collective and cyclic pitch angles. Whether this is a cantilevered pitch horn (the Triflex hub), a pitch cuff that encloses the flexbeam (the UH-60A tail rotor), or a torque tube that does not enclose the flexbeam (the BO 105/BMR), the primary purpose of the torque structure is to transmit a torsional moment to the blade root. Conceptually, the simplest possible torque structure is a pitch horn extending from the blade root to the pitch link which is very stiff in bending. In this case, a control input will twist the flexbeam but will also cause a flap deflection. Whether thought of as pitch washout or lost motion, the effect will be that for the same pitch link motion there will be less rotation of the blade root for this case than for a design with a pitch bearing. There will also be an effective pitch-flap coupling that will depend on the pitch link location. The only way in which a pure torsional moment can be transmitted to the blade root is if the pitch link load is reacted at the root of the torque structure with a shear restraint. This is the approach that has been taken with the UH-60A tail rotor and with the BO 105/BMR. Once a shear restraint is used at the torque structure root then a number of additional design options are opened. The torque structure itself no longer has to be stiff in flapping and chord, for it need only transmit the torsion moment, as on the BO 105/BMR. An elastomeric damper can be incorporated in the shear restraint to augment the lead-lag damping; however, to be effective it is necessary that there be significant relative motion between the root of the torque structure and the flexbeam. In this event, the torque structure must be very stiff in the chordwise direction. It is not clear that there is any advantage, however, in having the torque structure stiff in flapping, for this would tend to increase hub-moment stiffness. The effective flapping stiffness of the torque structure may be reduced, where it joins the blade root, by softening this joint so that it is effectively a pinned boundary condition.

The torque structure may be designed to enclose the flexbeam or to be separate from it. The decision to choose one option over the other may be made for aerodynamic reasons as much as for structural ones. An enclosing torque structure must be large enough to avoid contact with the flexbeam during normal twisting and bending. This will increase the hub frontal area and, hence, the hub drag, but with careful aerodynamic design of the torque structure or cuff it may be possible to reduce the drag over that of the flexbeam. If the torque structure is nonenclosing it may potentially be made smaller than a cuff design and, hence, have a reduced frontal area. An additional advantage of the nonenclosing torque structure is that the flexbeam and hub area are easier to inspect.

Design Approaches

Various design approaches are classified as to whether there is or is not a shear restraint at the root end of the torque structure, and if the torque structure carries torsion loads alone, or bending loads as well. The concepts proposed are shown in Fig. 12, using this scheme. Note that it does not appear feasible to design a torque structure that carries only torsion loads and is without a root shear-restraint. Most of the concepts proposed use both a root shear-restraint and an enclosing torque structure or cuff that carries bending loads as well, and most have the shear restraint centered at the root of the cuff. However, concept SA2 (Fig. 6) has a vertically offset shear restraint. The lower side of the torque structure is thickened so that its shear center passes through the offset restraint to avoid pitch-lag coupling. However, it does not appear that this would necessarily remove the coupling in this case.

An example of a design that uses a torque structure that does not carry bending loads is concept BH4 (Fig. 13). The torque structure is placed behind the flexbeam and lead-lag motion of the blade causes axial displacement of the torque tube with respect to the hub-mounted damper. Thus, elastomeric damping is incorporated in the design without the need to carry bending loads.

A curious result of the concept-definition studies is that despite past development of pitch-control concepts without a root shear-restraint, all of the present concepts use a root shear-restraint. In part, this is due to the additional swashplate travel that is required to compensate for the lost motion, and in part to some of the problems that have occurred in the previous development programs. Related to this is the inability of present analytical methods to satisfactorily predict the aeroelastic stability and flutter of these designs. In the development of the YUH-61A tail rotor, for instance, Boeing Vertol tested scale models of their design in hover and in the wind tunnel over a wide range of parameters.[12] They encountered flap-lag instabilities characteristic

of stiff inplane rotors in both the rotor cyclic and reactionless lead-lag modes, and flutter in both torsion and coupled flap-torsion modes. The flap-lag and flutter instabilities occurred for many different configurations and could not be adequately predicted by analysis. Eventually, a stable configuration was developed. In the case of the Triflex, no aeroelastic or flutter problems were reported, although from an aeromechanical stability point of view, the modal damping was less than desired and indicated that an improved elastomeric matrix was required.[12] Recent scale-model experiments with configurations without a root shear-restraint have also encountered a number of flutter instabilities,[23] and it is not known whether these can be predicted with current flutter analyses. The simplicity of the torque structure without shear-restraint is still very attractive, but it appears that our less-than-complete understanding of past development problems has resulted in a prejudice against this configuration that perhaps is undeserved.

Aeromechanical and Aeroelastic Stability

Design Considerations

In terms of aeromechanical stability (coupled rotor-fuselage stability) and aeroelastic stability of the isolated rotor, bearingless rotors behave very much like hingeless rotors. The major factors affecting hingeless-rotor stability (which apply to bearingless rotors as well) are discussed in Ref. 24. It seems likely that any bearingless main rotor that is developed will be a soft inplane rotor.[24] As a result, the critical stability problems will be aeromechanical, that is, ground and air resonance. Of major concern is the coupling of the rotor regressing lead-lag mode with a fuselage mode — normally the body pitch or roll mode. Under ground-contact conditions, the frequency and damping of the body pitch and roll modes will be determined in part by the landing gear characteristics, and to this extent they are under the control of the designer. As the helicopter lifts to a hover, the body frequencies and damping will change, and once airborne, the body frequencies will be determined by the fuselage inertias, something the designer has little control over. Throughout these regimes, it is desirable to maintain separation of the body frequencies and the regressing lead-lag mode, and to ensure that there is adequate damping in the rotor lead-lag mode.

There will be some structural damping in any flexbeam and blade design, usually of the order of 1 to 3% critical damping. This level of damping is probably not adequate for bearingless rotors throughout the entire flight envelope, and additional damping is likely required. This damping may be obtained from an auxiliary damping element incorporated in the flexbeam design or through aeroelastic couplings. An elastomeric damping element can sometimes be conveniently located in a hub design without a significant increase in hub drag; in general, however, there will be a detrimental effect on rotor weight, parts count, and MTBR. It is also possible to increase the rotor damping through the use of aeroelastic couplings, such as pitch-lag and structural flap-lag coupling,[25,26] and this may not necessarily impose a weight or drag penalty. Finding a suitable combination of couplings that is satisfactory over a range of operating conditions, however, remains a major design challenge. This approach requires valid mathematical models to support the aeroelastic design of bearingless rotors, and improvements in this area demand continued attention by the analyst.

Design Approaches Using Auxiliary Damping

Elastomeric materials may be usefully applied as auxiliary dampers since the additional 3 to 6% critical damping that is required for bearingless designs is well within the capability of elastomers. However, for the damping to be effective, the material must be significantly strained. Therefore, the elastomeric damper must be located where lead-lag motion will cause shearing across the elastomer. In present designs, such as the Model 680 main rotor, an elastomeric damper is incorporated in the root shear-restraint. Since the torque structure is stiff in chord bending, its root-end moves relative to the flexbeam shear-restraint mounting point during lead-lag oscillations, and the rotor damping is effectively augmented. This particular approach was used for a number of concepts (BH2, BH3, HH1, HH2, HH5, KA3, SA2, and SA5). The combination damper and root shear-restraint need not be located on the flexbeam, as is shown with concept BH4 (Fig. 13). In that case, the required shearing motion is due to axial motion of the torque structure with respect to the hub.

An alternative means of incorporating auxiliary damping is by adding a constrained layer of elastomeric damping material to the surface of the lag-torsion flexure. For this method to be effective, the elastomer must be backed or constrained by a very stiff material so that as the lag-torsion flexure bends, there is sufficient relative motion between the flexure surface and the backing material. This approach has been used by Boeing Vertol on

their BMR during wind-tunnel tests.[27] The damping augmentation by this method has varied between 1% and 4% critical damping, depending on operating condition.

Although not proposed in the studies, another approach to adding damping is to incorporate an elastomer in the flexbeam structure, as was done with the Triflex hub.[23] In that design, the flexbeam itself is made up of fiberglass cords in an elastomer matrix. However, it is not clear how much of the rotor damping is due to the elastomer.

Design Approaches Using Aeroelastic Couplings

The BO 105/BMR uses negative pitch-lag coupling that is obtained by coning the blade up 2.5° with respect to the flexbeam (negative droop), combined with structural flap-lag coupling obtained by pitching the flexbeam by 12.5°. Pitch-lag coupling and structural flap-lag coupling either separately or in combination may have beneficial stabilizing effects for aeromechanical stability. However, these effects do not hold true in a general sense, so that each new design must be carefully analyzed for stability and the results confirmed by model tests.

One approach to incorporating aeroelastic couplings is concept BH1, shown in Fig. 14. In this case, flap-lag coupling is incorporated in the flexbeam by rapidly transitioning the principal axes of the flexbeam from 0° at the hub to 15° at the shear-restraint. Pitch-lag coupling is not included in this design. Note that the torque structure encloses the flexbeam, even though it does not carry bending loads. This design illustrates some of the trade-offs that result in using aeroelastic couplings instead of auxiliary dampers. By eliminating the dampers, some bulk is removed at the shear-restraint, reducing the profile drag; but because the flexbeam is now pitched at 15°, the torque structure thickness is increased, which increases the profile drag. The rapid twisting of the flexbeam at its root is a trade-off between obtaining the required inclination of the principal axes and avoiding excessive fiber bending. The effective flap hinge can no longer be shifted inboard to reduce hub-moment stiffness without washing out the flap-lag coupling.

Concept BV1 (Fig. 8) is similar to the original BMR design in that negative droop is used to provide negative pitch-lag coupling. However, in this case, there is no pre-twist to the flexbeam to provide flap-lag coupling so the negative droop has been increased from -2.5° to -4°. An offset shear-restraint is used as with concept SA2; however, for this concept, the offset shear-restraint results in positive pitch-lag coupling which counteracts the negative droop. Concept BV2 is similar to BV1 except the flexbeam is made of two elements instead of four. In effect, the lower leading element and upper trailing element of BV1 have been removed, which provides structural flap-lag coupling. Concept BV3 is again similar to BV1, except 3° of precone are used instead of -4° droop, and the offset shear-restraint is shifted to the top of the hub where it provides negative pitch-lag coupling. Other mechanisms may be used to provide pitch-lag coupling. A variant of concept BH1 would use the axial motion of a nonenclosing torque structure (as in Fig. 13) to actuate a linkage that changes the blade pitch with lead-lag motion.

In studying possible variations of concept KA2 (Fig. 10), Kaman examined the effect of independently varying the vertical location and twist of the two elements of the flexbeam. In performing this study, they assume that the elements are cantilevered at the hub center. Clearly, if the leading element is offset vertically from the trailing element, this will provide flap-lag coupling, a configuration similar to BV2. However, from a practical point of view, this offset makes it difficult to carry the element from one side of the hub to the other, for a transition section is required to compensate for the offsets. What is not so obvious is that if the leading element is pitched up and the trailing element pitched down the resulting flexbeam will have negative pitch-lag coupling. This particular variation is practical in the sense that there is no difficulty in carrying the twisted element through the hub to the other side.

The various concepts presented for increasing stability through aeroelastic couplings all hold considerable potential for helping to deal with the complex aeromechanical stability design problem. However, the scope of these concept-definition studies was too limited to allow any stability analyses to support these conceptual design efforts, and it is essential that the most promising concepts be carefully analyzed for their aeromechanical and aeroelastic stability before proceeding with the preliminary design of the ITR/FRR rotor.

Materials for Bearingless Hub Designs

Although titanium was used for the flexural elements of the matched-stiffness, bearingless rotor for the XH-51A, the full potential of bearingless designs will only

be realized by taking advantage of modern composites. The specific tensile strength of composites is significantly greater than that of titanium, and the ability to orient the fibers in the matrix allows a torque structure to be torsionally stiff using ±45° fiber orientation; the torsion flexure made up of unidirectional fibers is torsionally soft as a result of the low shear modulus. However, despite their many attributes, the low shear strength of the matrix requires considerable care in designing with composites.

The three major composite formulations now used in the helicopter industry are made of fibers of graphite, fiberglass, or Kevlar in an epoxy matrix. The low density and high allowable strain of Kevlar make it a very attractive material; however, its very low compressive strength, approximately 20% of its tensile strength, makes it difficult to use in bearingless hub designs. Graphite-epoxy composites have high stiffness-to-weight ratios and good stress allowables, but poor fracture toughness and failure modes. Fiberglass, on the other hand, shows good strain allowables and good failure modes, but low stiffness-to-weight ratios.

A number of different approaches to selecting materials for a hub design were taken in the concept-definition studies. In Ref. 13, Boeing Vertol examined the trade-offs between graphite and fiberglass on the basis of a simplified flexure model that was optimized to minimize weight. For the same design constraints, they found the graphite flexbeam design lighter. Kaman addressed this question in a different way in Ref. 14 by looking at a materials merit-function based on tensile strength, bending stiffness, tensile-to-shear-modulus ratio, and density. On the basis of this merit function, they selected graphite for their concept KA2. They then made a trade-off study of the effect of substituting fiberglass for the flexbeam instead of graphite and found this increased the weight, flexbeam size, and torsional stiffness. In their optimization study for concept SA2, Sikorsky used both graphite and fiberglass as materials; as a result, the graphite flexbeam design was lighter with smaller dimensions and its hub moment-stiffness was 20% lower than for the fiberglass hub.

From these initial studies, it appears that graphite composites will provide a lighter weight design, less hub drag, and reduced hub-moment stiffness. But in terms of reliability and maintenance, the use of graphite presents some difficulties. Unlike fiberglass, in which crack propagation is slow, graphite shows degraded failure modes and may not be suitable for on-condition maintenance schemes. The final selection of a composite material may depend more on the size of the materials-allowable data base that exists within each company instead of on such readily measured quantities as specific tensile strength, stiffness modulus, or density. To the degree to which hub components are sized by material fatigue properties, the proprietary data bases and their associated knockdown factors for such effects as those of combined stresses and environmental conditions will define the material for best use. As experience is gained with present composites, and new composites and hybrid compositions are developed, the design difficulties should lessen in this area.

Other Design Considerations

Hub/Flexbeam Attachment

The hub and flexbeam attachment joint is a difficult and important design issue in a number of respects. Blade flapping and chordwise moments and centrifugal force are a maximum at the hub center and these loads must transition from the flexbeam to the hub and rotor shaft. To minimize the effects of centrifugal force on hub-moment stiffness, it is necessary to have the flexible portion of the flexbeam as close to the center of the hub as possible, and this minimizes the space available for transitioning the high bending and centrifugal loads. It is also desirable to keep the hub small in order to minimize weight and hub drag. Requirements for reduced vulnerability depend both on exposed area and on the ability of the structure to remain intact after a hit. A number of approaches have been used in the past to meet this variety of requirements, including building the hub and flexbeam up as a single unit (as was done with the 680 rotor), carrying the centrifugal loads through the hub by having opposite blade pairs constructed of a single flexbeam (as used on the UH-60A tail rotor) and bolting the flexbeam to the hub (as with the BO 105/BMR.

Single-Unit Flexbeam and Hub. The design approach taken on the Bell Helicopter Textron's concepts is to build the hub and flexbeam as a single unit or yoke (see Figs. 7, 13, and 14). In general, unidirectional fibers connect opposite blade pairs across the hub and are interleaved with cross-ply layers to build up the hub section. An upper hub plate is bolted through the composite hub to the upper end of the rotor shaft. Centrifugal loads are carried through the hub in the composite material; however, the rotor hub moment must be transferred to the hub clamping

plate without causing interlaminar shear failures over the transition region.

Interleaved Flexbeam Pairs. If opposite blade flexbeams are made as a single unit, they may be assembled as overlapping pairs (see Figs. 6, 8-10). The planes of the flexbeam pairs are then offset by the thickness of the flexbeam, but this is of small consequence. The hub is assembled in a manner similar to that of the single-unit hub, with an upper clamp plate bolted to the top of the rotor shaft. As with the single-unit hub, the rotor hub moment is transferred to the clamp plate over a short spanwise extent, and care must be taken in the design to avoid interlaminar shear failures.

For concepts BV1 to BV3, the primary rationale for going to stacked flexbeams was to reduce the interlaminar shear over the clamp radius. By dividing the single beam into two elements, the hub moment is divided into axial loads combined with smaller moments on each beam. The result is a considerable reduction in interlaminar shear.

Bolted Attachment. The flexbeam root-end may be connected to the hub with a bolted or pinned attachment. Although not shown here, this is the approach taken for concepts SA4 and SA5. For a bolted attachment it is necessary that the bending-moment distribution across the bolt not cause a load reversal and fretting. This tends to increase the height of the bolt and increases the hub's projected area and drag. An advantage of a bolted connection, however, is that aeroelastic couplings may be more easily incorporated in the design, for flexbeam pitch or coning angles are readily designed into the hub.

Flexbeam/Blade Attachment

Although conceptually attractive, the idea of a single rotor hub and blades or opposite blade pairs constructed as a single unit from tip to tip is not practical. Some means of blade-folding or of removing individual blades appears necessary in any design, and blade-folding must be addressed for the ITR/FRR. Normally, the inboard end of the blade and the outboard end of the flexbeam are designed so that the unidirectional fibers wrap around one or two bushings, and the flexbeam and blade are connected by one or more attachment bolts. It is not feasible to make a connection of this nature without increasing the bulk and drag at the attachment point. This point is far enough out on the blade radius to generate a nonnegligible rotor parasite power loss. The design challenge is how to minimize this drag increment while maintaining the load capacity of the attachment.

Droop Stops

As hub-moment stiffness is reduced, the flexbeam will no longer be able to carry the static droop loads during starting and stopping conditions, and the flexbeam will either have to be sized for static droop loads or droop stops will have to be added to the hub design. Because droop stops add [illegible] parts count and decrease MTBR they are [illegible] desirable, but they may be preferable [illegible] increasing the hub-moment stiffness, h[illegible]eight, and hub drag to accommodate sta[illegible] droop loads.

Concept SA2 (Fig. [illegible] shows the use of centrifugally actuated [illegible] stops. These are configured very muc[illegible] the droop stops used on current ar[illegible]culated-rotor hubs. Alternative approaches have been proposed to incorporate droop stops without centrifugal actuation. In concept BV1 (Fig. 8) the droop stop is part of the offset shear-restraint. As the blade droops down, a pin slides in a spherical bearing and stops against the hub. There is probably some weight and drag increase associated with strengthening the shear restraint and torque structure to take the droop loads, but this is not believed to be a significant problem. It is likely that the shear restraint for concept BV1 would be mounted on the top to provide the correct pitch-lag coupling, but the concept of a droop stop could be implemented in a similar fashion. Another approach is shown in Fig. 7 for concept BH2. In this case, the droop stop is simply an extension of the lower hub clamp plate that contacts the lower edge of the torque structure or pitch cuff. Additional weight would be required to strengthen these areas, but again, this does not appear to be a major problem. In both of these cases, it is expected that the increased hub-moment stiffness will result in a reduced flapping envelope relative to articulated rotor designs; therefore, centrifugally actuated droop or flapping stops are not required. However, as these designs are developed it will be necessary to ensure that the flight envelope is not limited by droop-stop pounding.

Comparison of Hub Concepts and ITR/FRR Technical Goals

After they had developed the 33 proposed hub concepts each of the five companies was asked to select at least two concepts for further development and to compare the attributes of those concepts with the ITR/FRR goals. The 13 concepts chosen for extended development and

comparisons with the goals are shown in Table 3. Because of the level of effort involved in the concept-definition studies, no calculations were made for MTBR and fatigue life. In addition, the torsional-stiffness goal which had been defined as "not requiring substantially more blade pitch control actuator force than required by current rotor system" was added to Table 2, for this will be a quantified goal in the next phase of the ITR/FRR Project.

In examining the performance of the selected concepts, it is necessary to keep in mind some of the limitations of the concept-definition studies. In general, the technical-goal definitions were not specific enough to provide an exact basis of comparison between the concepts. For example, in calculating the hub drag, the radial and vertical extents of the hub, its angle of attack, and how fuselage interference drag was to be treated were not defined. Similarly, for parts count, it was not specified what actually constituted the hub or what a standard part was. Another difficulty in comparing concepts is that the concept-definition studies established a means to trade off the goals (and some specification items) through a system of weighting functions, but this method turned out to be overly sensitive and relatively unsuccessful for these studies. However, some hub concepts were biased by the weighting functions used, and in this sense they are not representative of concepts that will be developed in the preliminary design phase.

The first goal, flat-plate area, shows a variation from 1.6 ft^2 for concept SA6 to 7.7 ft^2 for concept HH1. The large areas for concepts HH1 and HH8 reflect the emphasis placed on reducing the vulnerability of these configurations. The flat-plate area obtained for concepts BV1 to BV6 includes interference drag which was calculated to be 25% of the total hub drag. If this interference effect were removed, the drag areas would be more in line with those of the other configurations, which did not include interference drag. The low drag for concept SA6 reflects the low presented area of the elliptical-shaped hub.

The weight estimates ranged from 305 lb for concepts BV1 and BV3 to 656 lb for concept HH8. Again, the high values for concepts HH1 and HH8 are due to the emphasis on vulnerability. All other concepts indicate that the weight goal can be met by proper use of composite materials.

Parts count ranged from 29 for concept KA2 to 86 for concept BV4. A high parts count is generally typical of designs in which a shoe is used to alleviate [illegible] or of designs in which hinges are included. A significant number of the parts were required to meet the requirement for blade-folding. The original goal for a rotor-system parts count was 75, of which [illegible] were allocated to the hub. The design work performed under the concept-definition studies, plus an assessment of some recent bearingless designs, indicates that the current goal will not act as a technology driver and needs to be reduced. For the preliminary design phase this goal has been lowered to 50 parts for the rotor system.

The goals for hub-moment stiffness and endurance hub tilt-angle, as well as fatigue life, are closely coupled. In establishing the goals, it was desired to have a hinge offset of about 4.2%, assuming no stiffness in the flap bearing or flexbeam. The offset was purposely set low because of the effects of equivalent hinge offset for hingeless or bearingless designs on vibration, gust response, and handling qualities. The benefits of a high equivalent-hinge-offset, such as high control power, improved maneuverability, and increased center of gravity travel, must be traded off against detrimental characteristics, such as gust response, pitch-up instability, and, perhaps, vibration. The difficulty in setting a hub-moment stiffness goal is that the trade-offs involved are not well understood. From a structural and materials point of view, it is more difficult to obtain a low equivalent-hinge-offset design than a high one. Therefore, the goal of low hub-moment stiffness (2,600 ft·lb/deg) was established to push the design and manufacturing technology in this direction and thus provide future designers more freedom in choosing a particular level of hub stiffness. The hub-moment stiffness was originally established at an even lower value (1,750 ft·lb/deg); as work progressed, however, it became obvious that this goal was unrealistic, and it was raised to 2,600 ft·lb/deg.

The coupling of the hub-moment stiffness and endurance-limit flapping angle are shown in Fig. 15, where the values for the selected concepts are shown in a plot of endurance-limit flapping as a function of hub-moment stiffness. Concepts BV6 and SA1 use an elastomeric bearing for flap articulation; as a result, they show endurance-limit flapping of 8.0° and 8.57°, respectively, well above the goal of 5°. Concept HH8, which is a derivative of the AH-64A, uses a composite strap-pack and shoe and meets the goal, but probably could be tailored to provide additional endurance-limit flapping. Interestingly, concept BV5, which is classified as an

articulated configuration because of its lead-lag and pitch articulation, uses a flexbeam for flapping and is unable to meet either the hub-moment stiffness or endurance-limit flapping goal. The one gimballed configuration, concept SA6, meets both goals, largely because the gimbal spring is not centrifugally loaded and can be more easily tailored to meet the established goals.

The bearingless design options that use a shoe restraint, concepts BV4, HH1, and KA2, are able to tailor the flexbeam to meet the goals but impose some penalty in weight and drag because of the hub shoes. The classical bearingless designs cannot, in general, meet both goals. In this respect, it is noted that most of the studies examined endurance-limit flapping and the closely related fatigue life using simplified methods. Thus, for a particular design, bending stress and strain were calculated as functions of flapping angle; in some cases knockdown factors were taken for combined stress, and the results were compared with the company's materials-allowable data base, thus defining the endurance-limit flap angle. An engineering judgment was then made to determine whether the flapping beyond the endurance limit, which would occur during normal missions, would reduce the fatigue life below the goal of 10,000 hr. The exception to this is Ref. 16, where a prorated flapping spectrum based on UH-60A flight data was applied to the concepts. In the case of concept SA2, which had a calculated endurance-limit flapping angle of 3.28°, the calculation of fatigue life showed the design fell considerably short of the 10,000-hr goal. Concept SA2 was then modified to concept SA2a by adding material to the flexbeam until the 10,000-hr goal was achieved. This also improved endurance-limit flapping, but the hub-moment stiffness was increased even further beyond the goal. Even without considering the effect of fatigue life on the endurance-limit flapping and hub-moment stiffness goals, direct comparison of these concepts is difficult because each manufacturer used different knockdown factors and materials-allowables in defining endurance-limit flapping.

The concept-definition flap angle and stiffness goals appear too restrictive for bearingless designs, and for the ITR/FRR preliminary design these have been relaxed. A flapping spectrum has been defined for this phase, but the dependence of this spectrum, the 10,000-hr fatigue-life goal, and the endurance-limit flap angle are not well understood.

The torsional stiffness goal for preliminary design (200 in.·lb/deg) does not include the blade inertial and aerodynamic stiffening effects and therefore is not directly comparable to the values tabulated in Table 3 with the exception of the Boeing Vertol concepts. It appears, however, that this goal will be relatively easy to meet. Although none of the designs meets all of the goals, it appears that with careful selection of materials, tailoring of the geometry, and orientation of lay-ups that a bearingless design can come close to achieving the goals.

Concluding Remarks

Five concept-definition studies were conducted to examine hub concepts that could meet the technical goals of the ITR/FRR Project. Of 18 concepts examined, one was articulated, two used an articulated flap hinge with a lag-pitch flexure, four were gimballed, and 11 were bearingless designs. The bearingless design concepts have been examined in detail both to assess their potential for making significant advances in rotor technology and to show what major design problems remain. The three critical elements of bearingless rotor-hub design are 1) providing adequate flexbeam strength and fatigue life with low hub-moment stiffness, 2) reducing the weight and drag of the pitch-control mechanism, and 3) providing means of augmenting the rotor lead-lag damping to prevent aeromechanical instability.

Most of the concepts proposed showed some desirable attributes, and many have the potential to meet the ITR/FRR Project goals. In the area of flexbeam design, it appears that the individual designers have considerable latitude in their design approaches, and there are excellent opportunities for innovative design solutions. Both fiberglass and graphite composites were proposed, but when the two materials were compared for the same application, graphite was considered preferable because it reduced hub weight and size. However, the failure characteristics of graphite are inferior to those of fiberglass, and this remains a concern. In the area of material selection, hybrids or combinations of composites may prove superior to either parent.

Design options for the torque structure or pitch-control mechanism are more limited than for the flexbeam. All the concepts that were proposed included some form of shear restraint at the root of the torque structure and, in a sense, constitute a retreat from past efforts to design a torque structure without shear restraint (for instance, the YUH-61A tail rotor and the Triflex main rotor). Unlike its use

in most previous design efforts, however, the studies show that the shear restraint is being used here to introduce desirable couplings rather than to simply minimize modal couplings. In addition, many of the shear-restraint concepts proposed incorporate secondary functions, such as droop stops and elastomeric dampers, and in this respect they are significantly innovative.

Two primary methods are proposed for augmenting the rotor lead-lag damping - auxiliary dampers and aeroelastic couplings. Again, in most of the concepts proposed, some form of auxiliary elastomeric damper is used, a conservative approach to the problem. Aeroelastic coupling offers considerable potential for augmenting rotor damping, but requires a high level of confidence in the predictive capability of the aeroelastic mathematical model. However, even when auxiliary dampers are used, the need to minimize drag and weight will reduce elastomeric damper size, and this will require the use of valid mathematical models and confirmatory tests.

Although the concept-definition studies were limited in scope, they indicate that a number of the proposed concepts have the potential to meet the ITR/FRR technical goals. However, this conclusion must remain tentative until additional design work is performed in the next phase. The studies have also been useful in identifying areas in which there is some weakness in the design methods, and these will be emphasized in the preliminary design.

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Table 1. ITR/FRR hub technical goals.[a]

Rotor hub flat-plate drag area, ft^2	2.8
Hub weight as a percentage of design gross weight	2.5
Hub system parts count exclusive of standard fasteners	50
Hub-moment stiffness, ft·lb/deg	2,600
Maximum hub tilt-angle without fatigue damage, deg	5
Fatigue life, hr	10,000
Mean time between removal (MTBR), hr	3,000

[a]These goals are based on a design gross weight (DGW) of 16,000 lb and are scaled appropriately for higher DGW.

Table 2. Hub concepts.

Concept	Nomenclature	Reference	Figure (in noted reference)
BH1	Bearingless/damperless (concept No. 1)	12	20
BH2	Bearingless/outboard attachment (concept No. 2)	12	25
BH3	Bearingless/inboard blade attachment (concept No. 3)	12	5
BH4	Bearingless/short pitch-change element (concept No. 4)	12	6
BH5	Bearingless/gimbal (concept No. 5)	12	7
BH6	Bearingless/offset flap hinge (concept No. 6)	12	8
BH7	Pitch-change bearings/tension loaded flexure (concept No. 7)	12	9
BH8	Pitch-change bearings/unloaded flexure (concept No. 8)	12	10
BV1	Configuration 1A (modified BMR[a])	13	24
BV2	Configuration 1B (modified BMR[a])	13	25
BV3	Configuration 2B (advanced BMR[a])	13	26
BV4	Configuration 3 (flexure/shoe rotor)	13	27
BV5	Configuration 4 (reversed Starflex rotor)	13	28
BV6	Configuration 5 (lag/torsion flexure with flapping hinge)	13	29
HH1	Flat-strap cruciform (concept A)	14	32
HH2	Tapered cruciform (concept B)	14	2
HH3	Hemispherical flexbeam (concept C)	14	3
HH4	S-beam, laminated flexure (concept D-1)	14	5
HH5	S-beam, nonlaminated (concept D-2)	14	15
HH6	Multiple strap (concept E)	14	7
HH7	Flat-strap (concept F)	14	8
HH8	V-strap (concept G)	14	10
KA1	Plain elastic pitch beam (concept No. 1)	15	8
KA2	Plain elastic pitch beam (elastomer laminations) (concept No. 2)	15	19
KA3	Classic elastic pitch beam (concept No. 3)	15	11
KA4	Gimballed hub (concept No. 4)	15	14
KA5	Compound matrix pitch beam (concept No. 5)	15	16
SA1	Articulated composite-elastomeric head	16	3
SA2	Torque-tube/I-beam rotor	16	10
SA3	Soft-mounted rotor	16	12
SA4	C-beam rotor	16	13
SA5	Anvil-strap rotor	16	14
SA6	*Elastic gimbal rotor*	16	15

[a]Bearingless main rotor.

Table 3. Comparison of selected concepts and ITR/FRR technical goals.

Concept	Flat-plate drag area, ft^2	Hub weight, lb	Hub parts count	Hub-moment stiffness, ft·lb/deg	Endurance-limit flap angle, deg	Torsional stiffness, in.·lb/deg
BH1	2.31	386	34	5,875	5.37	570
BH2	2.43	400	38	3,165	3.96	490
BV1	3.54	305	47	4,190	3.4	146
BV3	3.54	305	47	4,190	3.4	146
BV4	5.04	478	86	1,500	5.0	450
BV5	3.96	436	67	5,810	3.2	150
BV6	5.43	461	85	1,830	8.0	300
HH1[a]	7.7	624	42	1,595	5.0	303
HH8[a]	6.7	656	78	2,050	5.0	298
KA2[a]	2.21	396	29	2,620	5.0	-
SA1	3.4	455	55	3,055	8.57	-
SA2	3.2	380	38	3,055	3.28	-
SA2a[b]	3.2	408	38	4,685	5.24	-
SA6	1.6	438	31	2,425	5.59	-
Concept-definition goal	2.8	400	50	2,620	5	-
Preliminary design goal	2.8	400	(c)	4,000	4	200

[a]Designed for 18,400 lb DGW, goals scaled to 16,000 lb.
[b]Stiffened version of SA2, see text.
[c]Rotor system parts count reduced from 75 to 50.

	ARTICULATED LEAD-LAG		FLEXBEAM LEAD-LAG	
	PITCH BEARING	PITCH FLEXBEAM	PITCH BEARING	PITCH FLEXBEAM
ARTICULATED FLAP HINGE	SA1 UH-60A			BH6 BV6 206 T/R
GIMBALLED FLAP HINGE				BH5 KA4 SA3, SA6
FLEXBEAM FLAP HINGE	BH7, BH8 BV5 HH8 AH-64A BHT 412 STARFLEX	HH6	BO 105 YUH-61A	BH1–BH4 BV1–BV4 HH1–HH5, HH7 KA1–KA3, KA5 SA2, SA4, SA5 UH-60A T/R BO 105/BMR

Fig. 1. Classification of hub concepts by the means of providing flap, lead-lag, and torsional flexibility. Current designs shown in boxes.

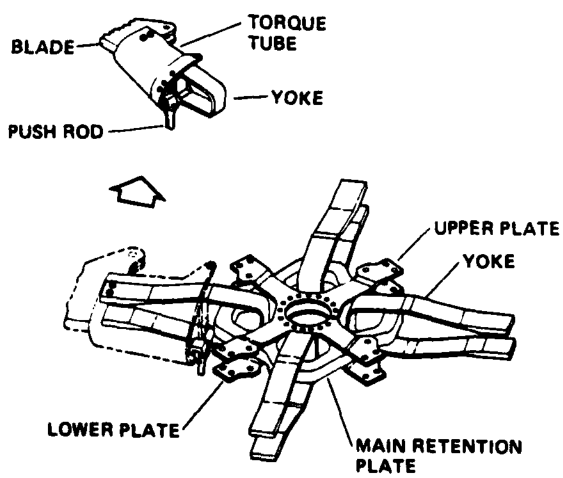

Fig. 2. Concept SA1, articulated hub made of composite materials and using elastomeric bearings (Ref. 16).

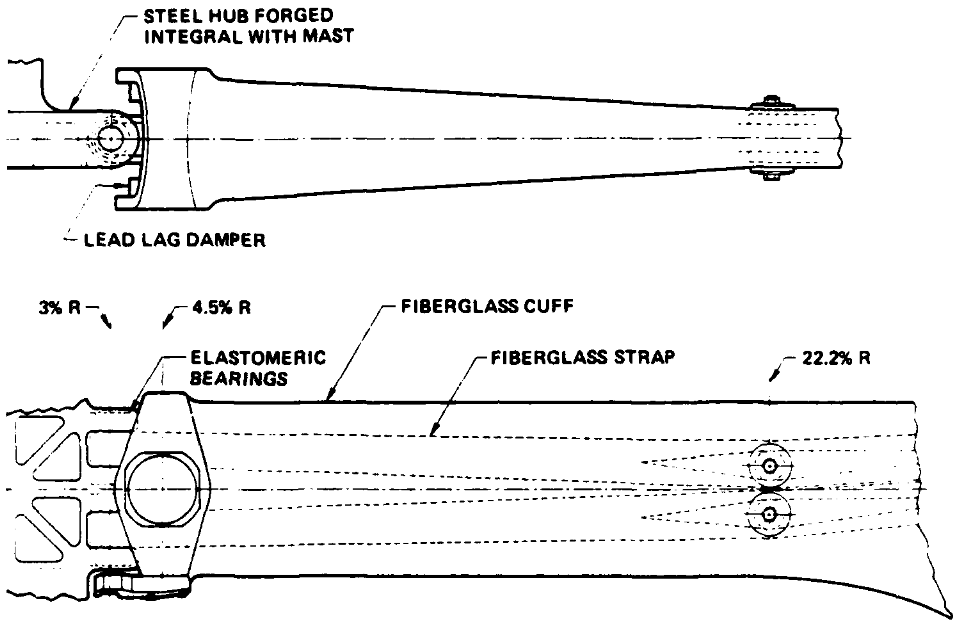

Fig. 3. Concept BH6, lag-torsion flexure combined with articulated flapping hinge (Ref. 12).

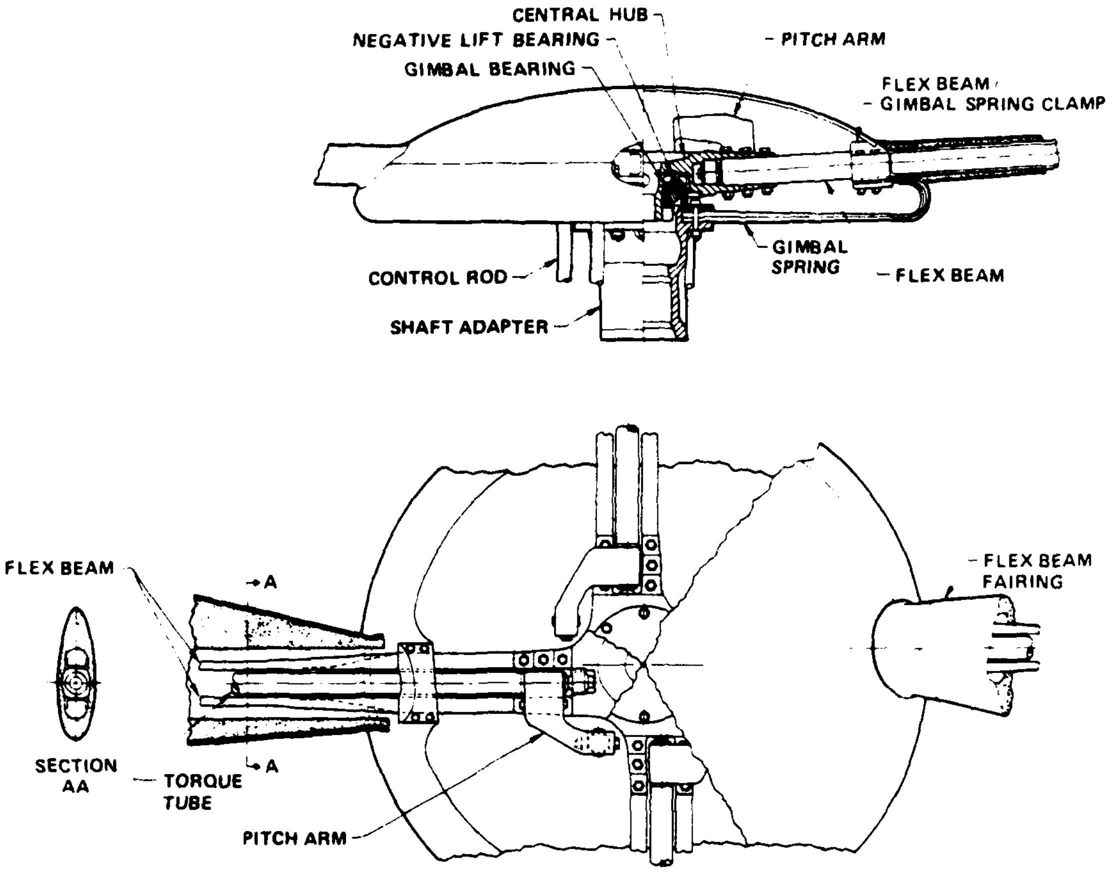

Fig. 4. Concept SA6, lag-torsion flexure combined with elastomeric gimbal and spring (Ref. 16).

		SINGLE ELEMENT	DUAL/QUAD ELEMENT	LAMINATED ELEMENT
UNTAILORED FLEXBEAM	NO SHOE	HH5, HH7 KA5 SA2, SA4 UH-60A T/R TRIFLEX	BV1–BV3 HH3 BO 105/BMR	BH4
	SHOE		KA1, KA3 SA5	KA2
TAILORED FLEXBEAM	NO SHOE	BH1, BH2 BHT 680	BH3	
	SHOE	BV4		HH1, HH2 HH4

Fig. 5. Classification of flexbeam design concepts on the basis of flexbeam tailoring, the use of a shoe to control curvature, and whether the flexbeam is single-element, multielement, or laminated element. Current designs shown in boxes.

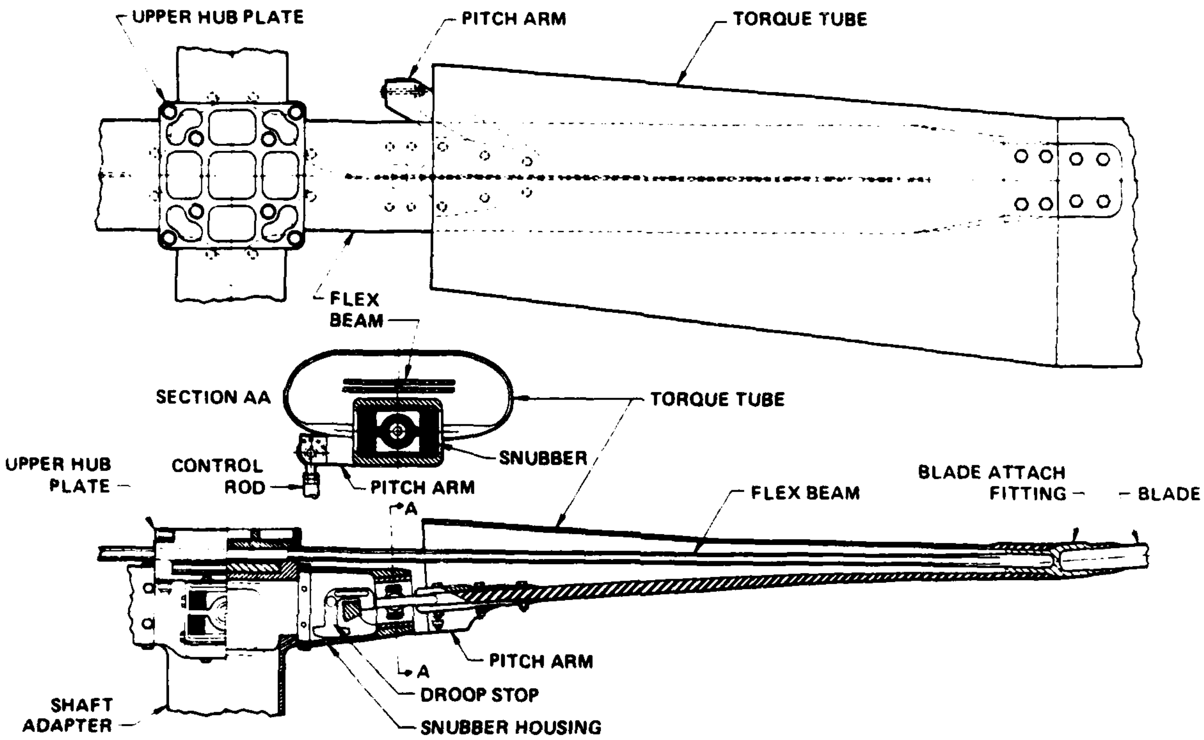

Fig. 6. Concept SA2, flexbeam with graphite I-beam cross section (Ref. 16).

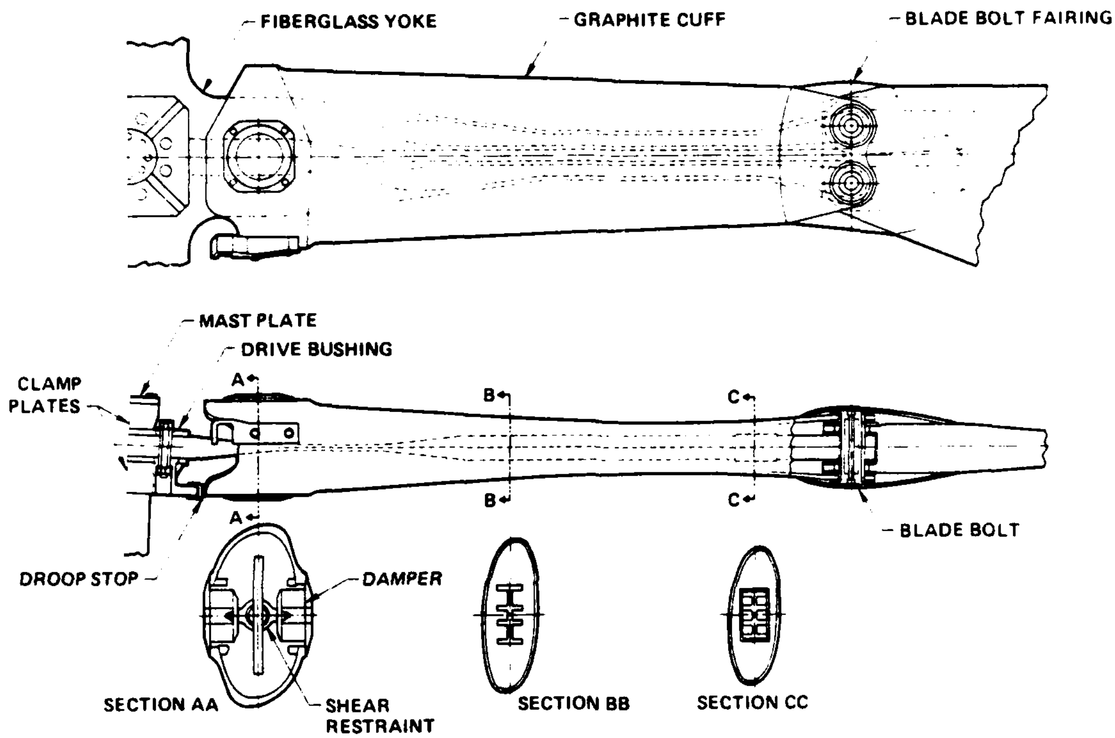

Fig. 7. Concept BH2, tailored fiberglass flexbeam (Ref. 12).

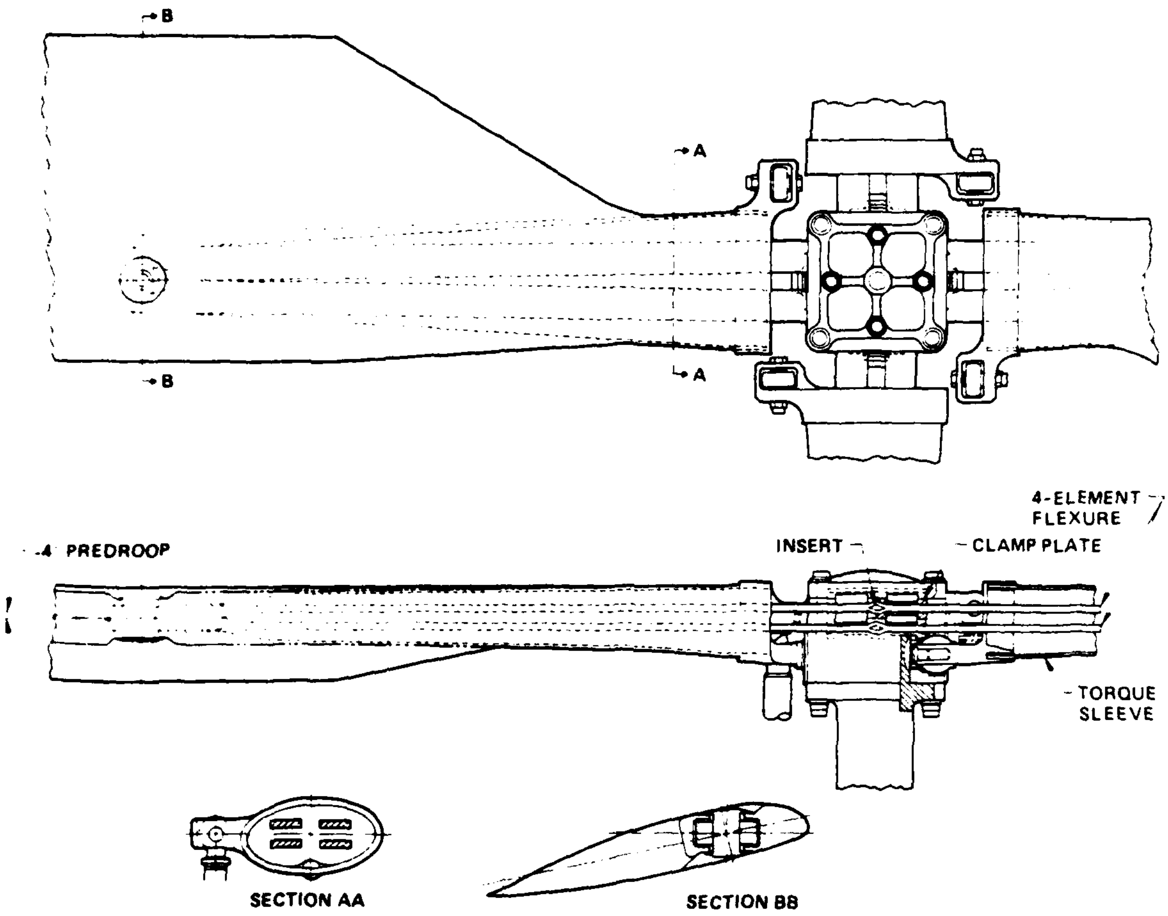

Fig. 8. Concept BV1, four-element flexbeam (Ref. 13).

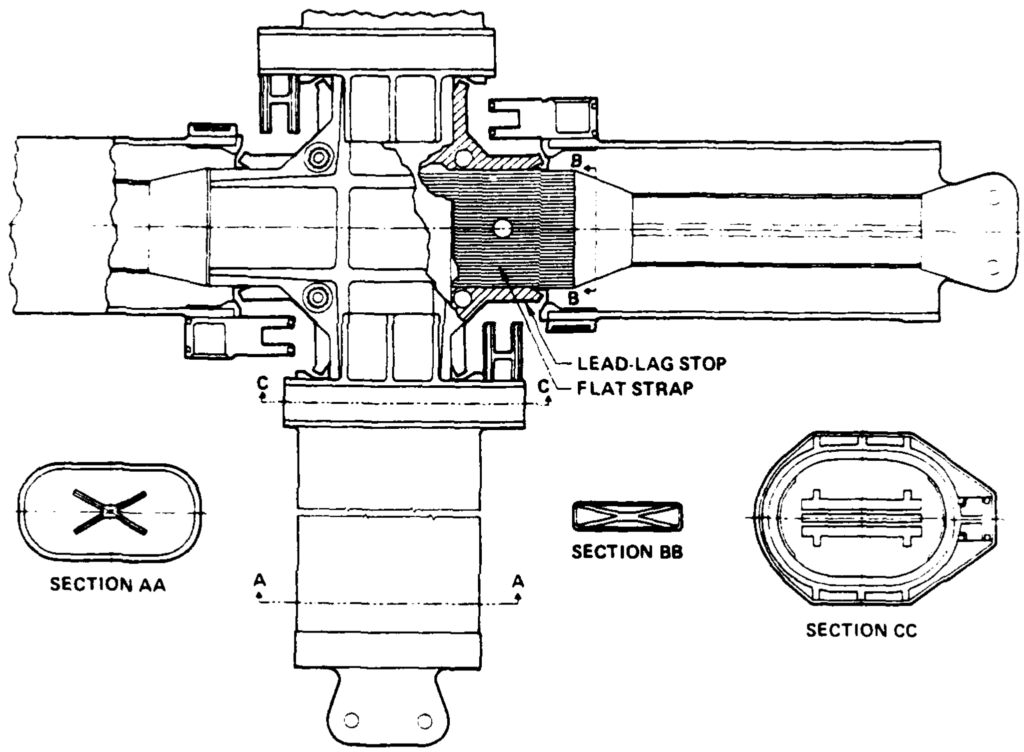

Fig. 9. Concept HH1, shoe-restrained, laminated fiberglass flap flexure transitioning to a flattened X lag-torsion flexure (Ref. 14).

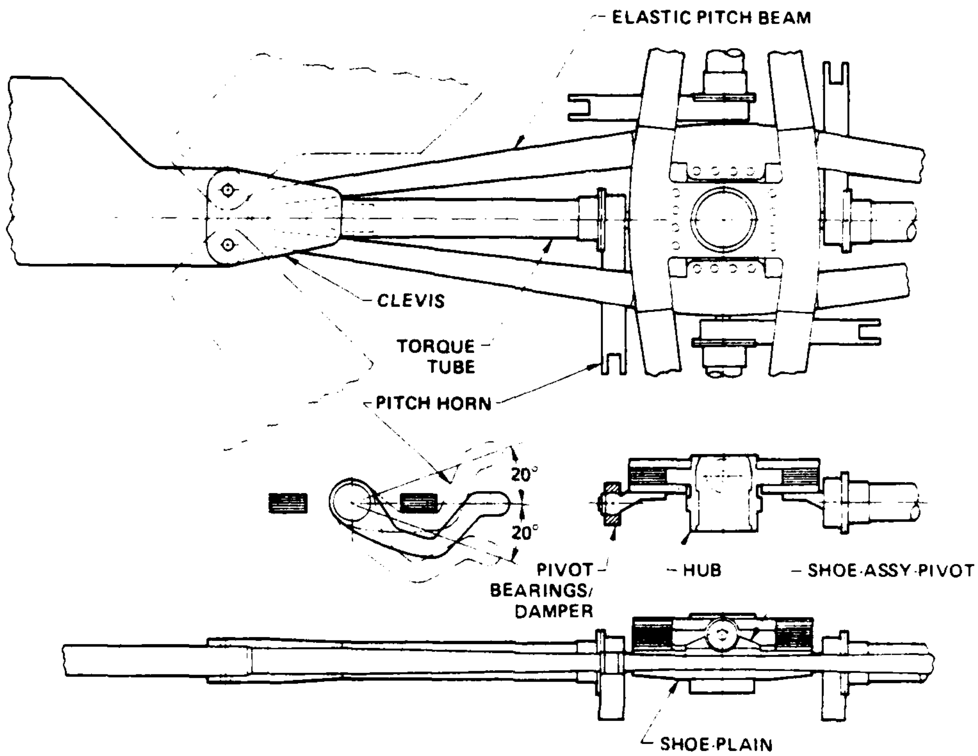

Fig. 10. Concept KA2, dual-element, laminated graphite flexbeam (Ref. 15).

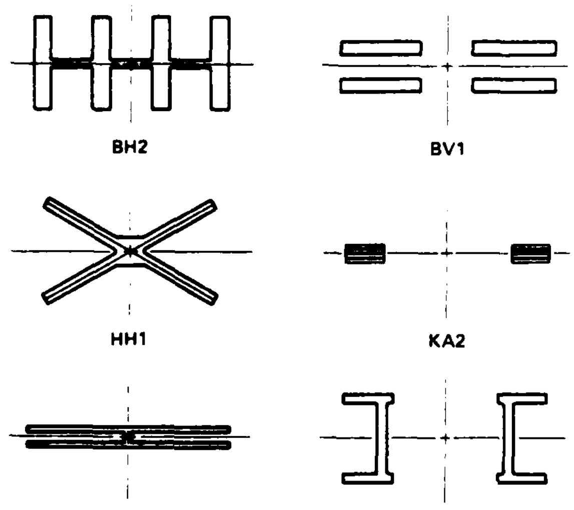

Fig. 11. Cross sections of lag-torsion flexures of various concepts.

	TORSION LOADS	TORSION AND BENDING LOADS
ROOT SHEAR RESTRAINT	BH1, BH4 KA1 SA4 BO 105/BMR	BH2, BH3 BV1–BV4 HH1, HH2, HH6 KA2, KA3 SA2, SA5 BHT 680 UH-60A T/R AH-64A CFTR
NO ROOT SHEAR RESTRAINT	NOT FEASIBLE	TRIFLEX YUH-61A T/R BK 117 T/R

Fig. 12. Classification of pitch-control concepts by use of root shear-restraint and load capacity of torque structure. Current designs shown in boxes.

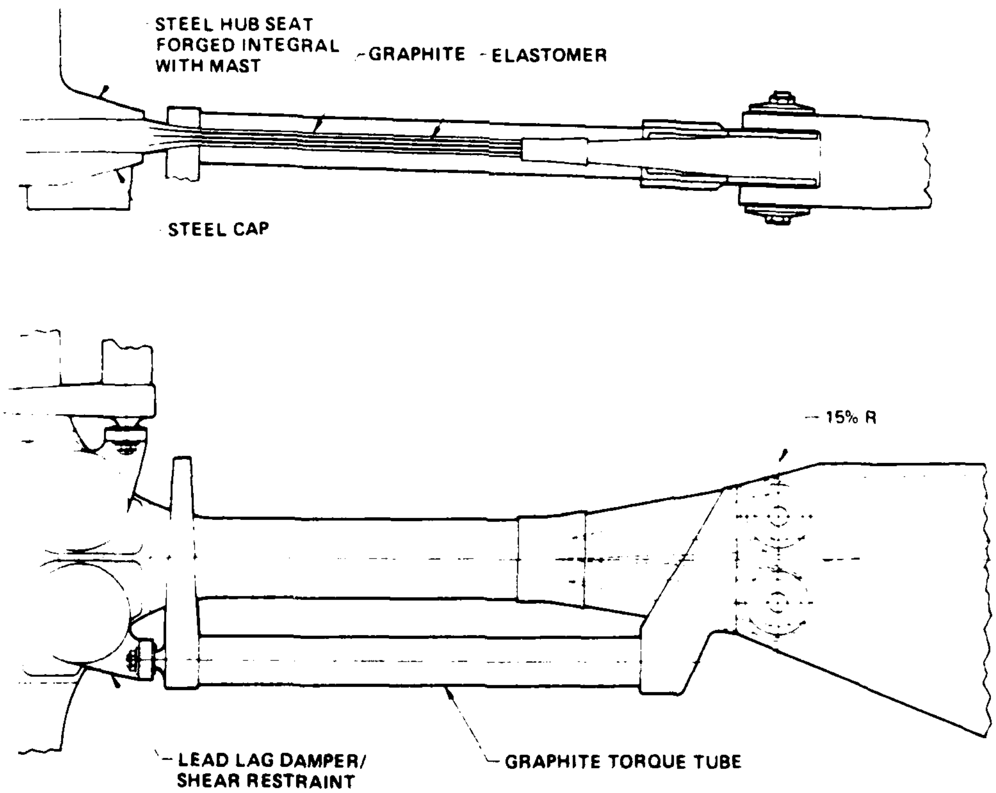

Fig. 13. Concept BH4, laminated graphite flexbeam with nonenclosing torque structure (Ref. 12).

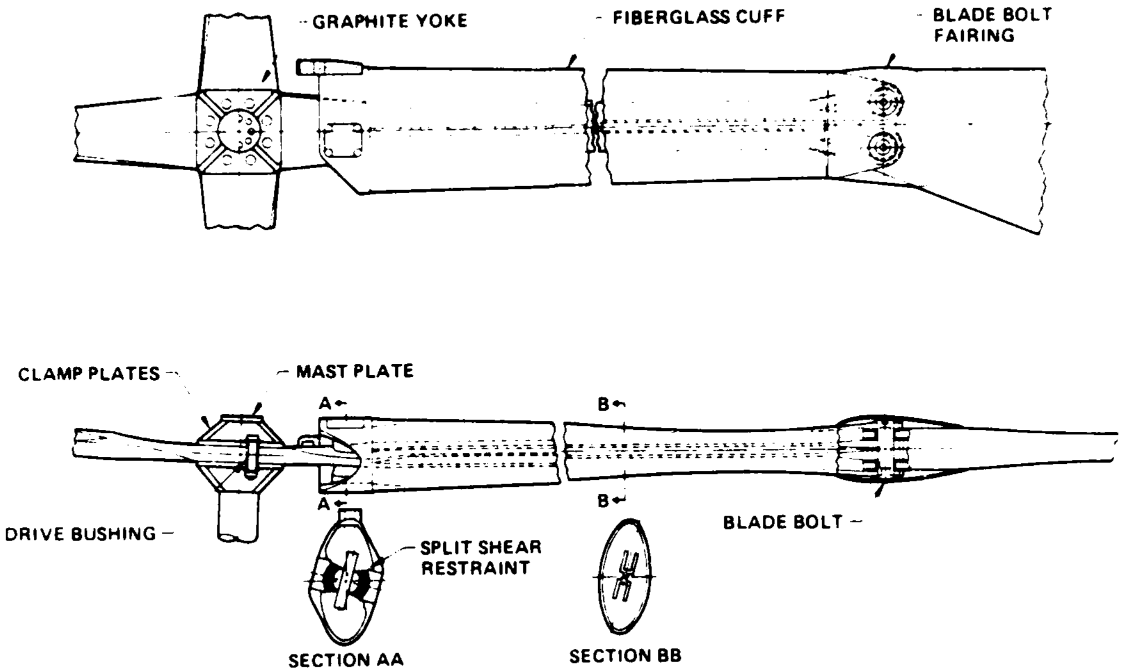

Fig. 14. Concept BH1, structural flap-lag coupling in graphite flexbeam design (Ref. 12).

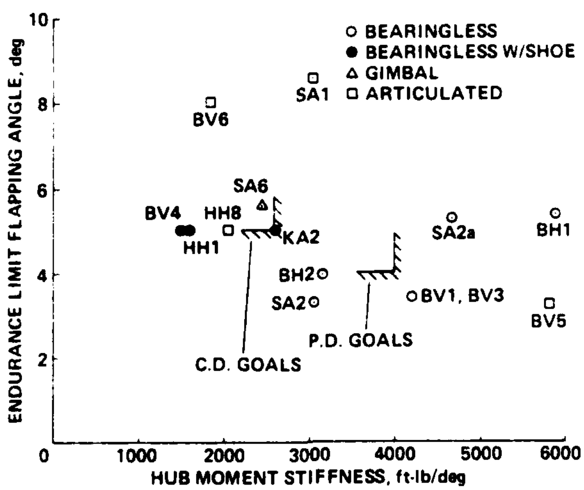

Fig. 15. Endurance-limit flapping angle as a function of hub-moment stiffness for selected concepts.